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Economic considerations and cost comparisons between the heat pumps and solar collectors for the application of plume control from wet cooling towers of commercial buildings

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Abstract

This communication presents a case study based on the economic considerations and comparisons between the heat pump and solar collector heating systems for the application and utility to control the visible plume from wet cooling towers of a huge commercial building in Hong Kong. A detail economic study for both cases, i.e. for heat pumps as well as for solar collectors is done and compared using different (capital and operational) costs, taking other constraints into account. The capital cost is the actual cost of the device, for example, for a heat pump it is the cost of the heat pump machine. For a solar collector it is the cost of all the components like the collector, pipes, pump, heat exchanger, etc. On the other hand, the operational cost is the cost that keeps the system working in good condition. For a heat pump, the cost of the input power to the compressor is the running cost, while the necessary maintenance and replacement of parts comes under other cost. Similarly, for a solar collector, the cost of the power consumed by the pump/compressor to circulate the working fluid is the running cost which is very less as compared to the former. It is found that all the costs are much lesser for a solar collector system while it is reverse in the case of an air-cooled

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geothermal heat pump system. Other comparisons between the electric and geothermal heat pump systems are also given among different possible options.

Keywords: Wet cooling tower; Solar collector; Heat pump; Water-cooled chiller; Visible plume; Commercial building; Ambient air temperature; Relative humidity

Contents

1. Introduction

Cooling towers are the enclosed boxes for the evaporating cooling of water by direct contact with the air, which is achieved partly by an exchange of latent heat from the water evaporation and partly by a transfer of sensible heat. They are used for heat rejection purposes and are among the important component of any power plant, HVAC, and industrial applications, especially, where the availability of cooling water is not sufficient. Physically they are often the single largest item of plant at the site and their visual impact is greatly increased if they emit a visible plume of superheated air into the atmosphere. The development of the cooling tower analysis began in the 19th century with the first work carried out by Lewis [1] and was used by Robinson [2] for the first time to establish the general principles, applicable to cooling tower design and derived equations for the designers. He further stated a series of fundamental concepts about the mechanism involved in the transfer of heat between liquid, gasses, and on the vaporization of liquid. Merkel [3] used the enthalpy potential as the driving force for air–water exchange and assumed a similarity between heat and mass convective transfer by means of Lewis number equals to unity which has been used to date.

For several decades, many authors have studied the convection phenomena occurring in cooling towers. Baker and Shrylock [4] developed a detailed explanation of the concept of cooling tower's performance clarifying the assumptions and approximations used by Merkel. Sutherland [5] showed that Merkel's theory leads to an underestimation of tower size by 5–15%. Braun [6] developed a confined method to model the performance of both cooling tower and dehumidifying coils. Based on Merkel's theory, the effectiveness-NTU relationships method has been developed by Braun et al. [7], taking into account the saturated air specific heat used for sensible heat exchanger. In the model, two parameters,

air-side and water-side heat transfer coefficients, were introduced. The results of this method were compared with numerical solutions of the detailed heat and mass transfer models and experimental results.

In order to investigate the effect of (low) water and air flow rates, Shelton and Weber [8] used a mathematical model based on the performance data of the manufacturer. Lebrun [9] presented the fundamentals of a new simulation toolkit oriented upward simple solutions for primary HVAC equipments. This toolkit used as much as possible models and each model is described by reference to a conceptual scheme, in which the very classical engineering components are interconnected. Bernier [10] presented a unidimensional analysis of the spray type counter flow cooling tower showing the influence of fill height, water retention time, and water flow rate on the performance of the cooling tower. Lebrun and Silva [11] presented an analysis of basic heat and mass transfer processes occurring around a droplet in transient cooling tower. Some study on heat and mass transfer analysis on the cooling tower were also made by Naphon [12], Kloppers and Kröger [13–15], Lees [16], and Smith [17] using different methods.

During unfavorable weather conditions, the exhaust of the cooling tower remixes with the cooler ambient air and as it cools down the excess moisture condenses in small fog droplets, creating visible plume. Nowadays, the visible plume from huge buildings attracts the public attention and is a matter of greater concern at places like Hong Kong. Under the most unfavorable combination of the ambient conditions, thermal load and topography, such a plume can extend up to few hundred meters and sometime causes visibility and darkness [18–24].

Methods of reducing (preventing and/or removing) the visible plume have taken many forms, such as heating the exhaust with natural gas burner, steam coils, installing precipitators, and spraying chemicals on the tower exhaust. However, such types of solutions, in general, are expensive and are not always effective [18,23]. A little can be done in a wet cooling tower to reduce/remove plume persistence. It is observed that the condition for the visible plume is the lower ambient temperature and the higher relative humidity. The latest literature review about different cooling towers shows different options and methods for reducing the visible plume from cooling towers, such as the of combination of wet–dry cooling towers, hybrid cooling towers, dry cooling towers, and so on, depending upon the need and demand.

As mentioned by number of researchers, wet cooling towers are economically cheap and technically simple, easy to build, and need less power to operate as compared to other types of towers. But they do not have any control over the visible plume [18–24]. In recent years, the visible impact of releases to the ambient has become a matter of greater concern due to the awareness of environmental degradation and protection among the society. Very recently, some authors [26–28] used the heating and cooling strategy by heat pumps [26,27] and solar collectors [28] to control the visible plume from wet cooling towers.

The present work is a comparative economical study on the application and utility of the heat pumps and flat plate solar collectors [29] heating system to control the visible plume from wet cooling towers of a huge commercial building in Hong Kong following earlier workers [26–28]. A detail economic study for both cases, i.e. for heat pump as well as for solar collector is done and compared using the investment cost, and operational cost, taking the other constraints into account. In any of the previous cases [26–28] the economic consideration which is a very important part of a real-life problem was not

considered. The comparison is also given among the different possible options for different combinations given earlier [26–28].

2. System description and analysis

There is a central air conditioning system, containing 6 water-cooled chillers and 10 wet cooling towers, besides other equipments such as the heat exchanges, connecting pipes, pumps, etc. Since the schematic diagram of the entire system is too complex to design in a compact figure, so a simple line diagram is given in Fig. 1(a). The complete line diagram [Fig. 1(a)] consists of a chiller coupled with a building envelope (space to be cooled), a cooling tower (to extract heat from the chiller), and a heat pump/solar collector (to heat the exhaust of the cooling tower). As can seen from Fig. 1(a), the refrigerant absorbs the

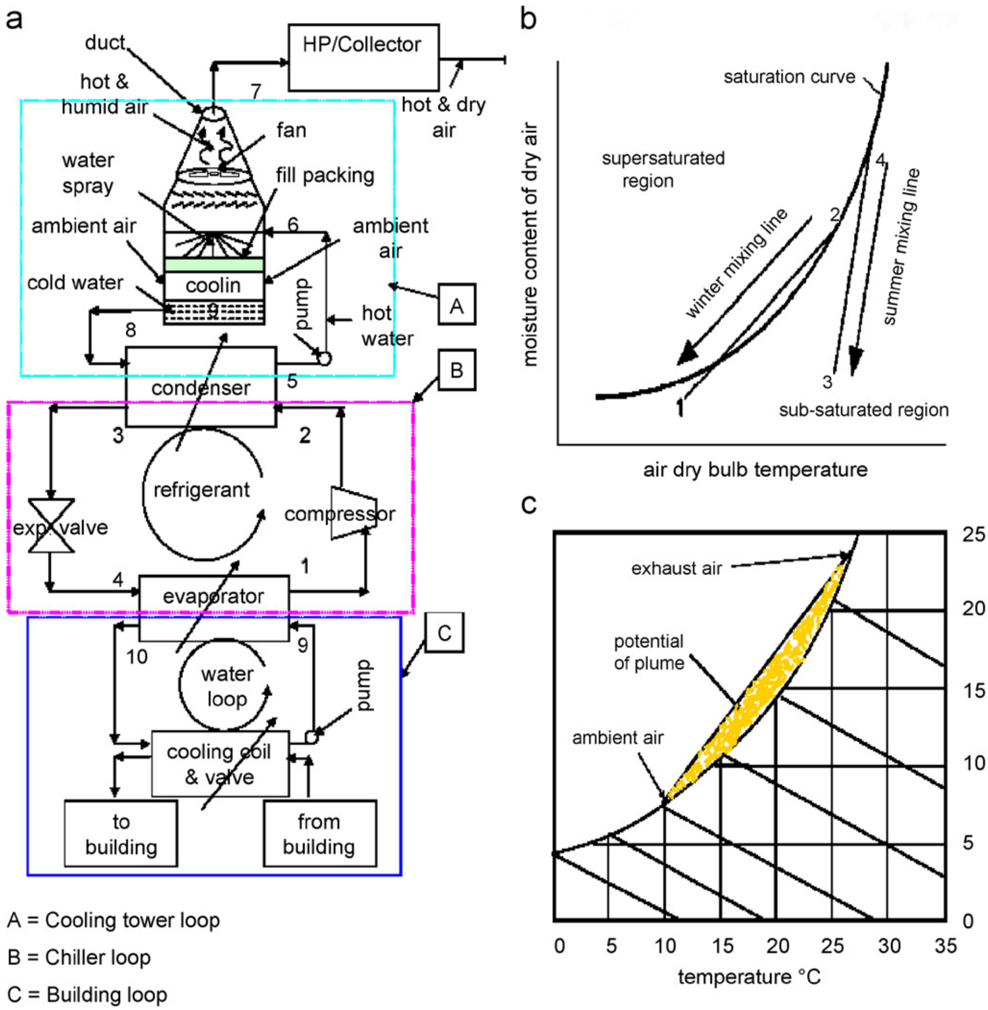

Fig. 1. Line diagrams of (a) the proposed system, (b) the formation of plume (white fog) in a wet cooling tower, and (c) the potential of plume in a wet cooling tower at a typical weather condition in Hong Kong.

heat from the evaporator and changes from saturated liquid state to saturated vapor state and enter the compressor, where it is compressed up to (superheated) state point 2. At state point 2 the refrigerant enters the condenser and cools down up to state point 3 (saturated liquid) by rejecting the heat to the circulating water from the cooling tower. At state point 3, it enters the expansion valve and being expanded up to state point 4 and again enters the evaporator, thereby, completing the chiller sub-system.

The water entering the condenser at state point 8 absorbs the heat from the refrigerant rejected in the condenser and then pumped back to the cooling tower at state point 6. The hot water at state point 6 is sprayed over the fill packing, and comes in the contact of the ambient air being circulated by the exhaust fan on the top of the cooling tower shown in Fig. 1(a). The hot water is cooled down up to state point 8 and the heat is taken away by the ambient air through evaporative cooling. The air entering the cooling tower through side openings leaves the tower through the exhaust fan via a duct at state point 7, completing the cooling tower sub-system. On the other hand, the circulating water leaving the evaporator at state point 10 enters the cooling coil where the secondary fluid is circulated to cool down the building space and the higher temperature water is pumped back to the evaporator at state point 9, thereby, completing the building sub-system. The analysis of main sub-systems viz. the chiller and cooling tower along with the heating and cooling requirements (in brief) is given below.

2.1. Chiller simulation

The chiller model simulates the chiller performance under various working conditions on the base of impeller tip speed (u_2), impeller exhaust area (A), impeller blade angle (β), and 13 other coefficients/constants. This might be available from chiller technical data and/or can be identified (partially or fully) by an associated pre-processor using chiller performance data under full load and partial load. The chiller chilling capacity is assumed to be controlled by adjusting the inlet vane angle (θ), the refrigeration cycle of a two-stage centrifugal chiller model was given by Wang et al. [25]. The compressor is modeled on the basis of mass conservation, Euler turbo-machine, and energy balance equation. The Euler equation is modified by considering the impeller exit radial velocity (c_{r2}) distribution and derived as:

$$h_{th} = u_2\left[u_2 - \left(\frac{\pi^2}{8}\right)^2 c_{r2}\left(ctg\beta + B\frac{v_1}{v_i}tg\theta\right)\right], \tag{1}$$

where h_{th} is theoretical head, B is the ratio of impeller channel depth at intake to that at exhaust, v_1 and v_i are specific volume at impeller intake and exhaust, respectively. Energy balance equations are applied on two control volumes, i.e. compressor control volume (from compressor suction to compressor exhaust) and impeller control volume (from compressor suction to impeller exit) as given by the following equations:

$$h_{th} = h_{pol.comp} + h_{hyd.comp}, \tag{2}$$

$$h_{th} = h_{pol.imp} + h_{hyd.imp} + \frac{c_i^2}{2}, \tag{3}$$

where h_{pol} is polytrophic compression work, h_{hyd} is hydrodynamic losses, c_i is the vapor velocity at impeller exhaust. The hydrodynamic losses in two control volumes are

considered to be composed of three elements (i.e. flow friction losses, inlet losses and incidence losses) as given by the following equations:

$$h_{\text{hyd.comp}} = \varsigma\left[1 + \psi_1\left(\frac{v_1}{v_i}\frac{1}{\cos\theta}\right)^2 + \psi_2\left(\frac{v_1}{v_i}tg\theta\right)^2\right]c_{\text{r2}}^2, \tag{4}$$

$$h_{\text{hyd.imp}} = \varsigma\left[\chi + \psi_1\left(\frac{v_1}{v_i}\frac{1}{\cos\theta}\right)^2 + \psi_2\left(\frac{v_1}{v_i}tg\theta\right)^2\right]c_{\text{r2}}^2, \tag{5}$$

where ζ, ψ_1, ψ_2, χ are the introduced constants. The flow friction losses are considered to be proportional to the square of the compressor volume flow rate and hence, proportional to the square of the impeller exit radial velocity (c_{r2}^2). The inlet losses are considered to be proportional to the square of the velocity through pre-rotation vanes channel. The incidence losses are considered to be proportional to the square of the shock velocity component. Given the evaporator pressure, condenser pressure, and position of inlet vanes (value of θ), the compressor model can calculate radial velocity and specific volume at impeller exhaust, and hence, the refrigerant mass flow rate and internal power consumption. The compressor capacity is controlled by the inlet vane angle (θ) as given in Eqs. (4) and (5).

For multi-stage chillers, single-stage compressor equations are used to calculate the first stage. The second stage is assumed to have the same flow, efficiency, and compression ratio as the first stage. Mass and energy conservations are applied to the economizer and the mixing process at the second stage suction. Only the first stage impeller geometric parameters are of concern. The evaporator and the condenser are simulated using the classical heat exchanger efficiency method. By considering the effects of water flow rate ($M_{\text{w,ev}}$, $M_{\text{w,cd}}$) and heat flux (Q_{ev}, Q_{cd}), the evaporator and condenser overall heat transfer coefficient–area products (UA_{ev}, UA_{cd}) are represented empirically, by the following equations:

$$UA_{\text{ev}} = [C_1 M_{\text{w,ev}}^{-0.8} + C_2 Q_{\text{ev}}^{-0.745} + C_3]^{-1}, \tag{6}$$

$$UA_{\text{cd}} = [C_4 M_{\text{w,cd}}^{-0.8} + C_5 Q_{\text{cd}}^{1/3} + C_6]^{-1}, \tag{7}$$

where C_1–C_6 are constants, the evaporation and condensation temperatures, and hence, the evaporator and condenser pressures are calculated, for the given chiller capacity (Q_{ev}), heat rejection (Q_{cd}), chilled and cooling water flow rates and inlet temperatures.

The power consumption of the chiller (W) is calculated on the basis of the compressor internal power consumption (W_{in}). With the consideration of mechanical and leakage losses of the compressor, electrical and mechanical losses of the motor, the chiller power consumption, can be calculated by the following equation:

$$W = \alpha \cdot W_{\text{in}} + W_1, \tag{8}$$

where α is a coefficient.

2.2. Cooling tower simulation

The cooling tower model is used to simulate the states of outlet air and outlet water of the cooling tower. In this simulation, the effectiveness model for cooling towers developed

by Braun et al. [6] is used. Based on the steady-state energy and mass balance of an incremental volume, the following differential equations can be derived:

$$\mathrm{d}m_\mathrm{w} = m_\mathrm{a}\,\mathrm{d}\omega_\mathrm{a}, \tag{9}$$

$$\frac{\mathrm{d}T_\mathrm{w}}{\mathrm{d}V} = \frac{\dfrac{\mathrm{d}h_\mathrm{a}}{\mathrm{d}V} - C_\mathrm{pw}(T_\mathrm{w} - T_\mathrm{ref})\dfrac{\mathrm{d}\omega_\mathrm{a}}{\mathrm{d}V}}{\left[\dfrac{m_\mathrm{w,i}}{m_\mathrm{a}} - (\omega_\mathrm{a,o} - \omega_\mathrm{a})\right]C_\mathrm{pw}}, \tag{10}$$

where m_w is the mass flow rate of water, m_a is the mass flow rate of dry air, ω_a is the air humidity ratio, h_a is the enthalpy of moist air per mass of dry air, C_pw is the specific heat of water at constant pressure, T_ref is the reference temperature for zero enthalpy of liquid water. The effectiveness of cooling tower is used to simulate the heat and mass transfer processes in the cooling tower as given in the following equation:

$$Q = \varepsilon_\mathrm{a} m_\mathrm{a}(h_\mathrm{s,w,i} - h_\mathrm{a,i}), \tag{11}$$

where ε_a is the air side heat transfer effectiveness, $h_\mathrm{a,i}$ is the enthalpy of inlet moist air per mass of dry air, $h_\mathrm{s,w,i}$ is the saturation air enthalpy with respect to the inlet temperature of water surface. The outlet air state and water state can be determined through energy balance as given by the following equations:

$$h_\mathrm{a,o} = h_\mathrm{a,i} + \varepsilon_\mathrm{a}(h_\mathrm{s,w,i} - h_\mathrm{a,i}), \tag{12}$$

$$\omega_\mathrm{a,o} = \omega_\mathrm{s,w,e} + (\omega_\mathrm{a,i} - \omega_\mathrm{s,w,o})\mathrm{EXP}(-\mathrm{NTU}), \tag{13}$$

$$T_\mathrm{w,o} = T_\mathrm{ref} + \frac{m_\mathrm{w,i}(T_\mathrm{w,i} - T_\mathrm{ref})C_\mathrm{pw} - m_\mathrm{a}(h_\mathrm{a,o} - h_\mathrm{a,i})}{m_\mathrm{w,o}C_\mathrm{pw}}. \tag{14}$$

2.3. Requirements of heating and cooling

For a typical weather conditions, the formation and the potential of visible plume are shown in Figs. 1(b) and (c), while the hourly ambient air temperature and relative humidity are shown in Fig. 2. Since, the present method is a technique to cool and then heat and/or to heat the exhaust of the cooling tower, the requirements of heating and cooling as well as heating are shown in Figs. 3(a) and (b), respectively. This is clear from these figures that the heating is carried out in a way the exhaust of the tower while mixing with the ambient air along the line B to C remains in the sub-saturated region as also mentioned in Fig. 1(b). The straight line AB [Figs. 3(a) and (b)] shows the requirements of heating, while the line AC through the saturation curve shows the cooling pattern. The heating and cooling requirements are given by the following equations:

$$Q_\mathrm{H} = \dot{m}_\mathrm{a} C_\mathrm{pa}(T_\mathrm{aa} - T_\mathrm{set}), \tag{15}$$

$$Q_\mathrm{L} = \dot{m}_\mathrm{a} C_\mathrm{pa}(T_\mathrm{set} - T_\mathrm{aset}), \tag{16}$$

where $\dot{m}_\mathrm{a}$ and C_pa are, respectively, the mass rate and specific heat of the tower exhaust air, while T_aa, T_set, and T_aset are, respectively, the outlet, set point, and the adjusting set point temperatures of the exhaust air to be heated and/or to be cooled down in order to control the visible plume. All other parameters can be measured directly from the system and/or

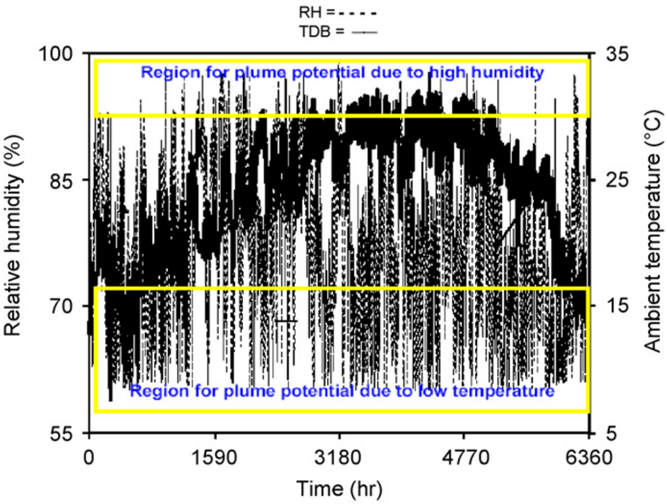

Fig. 2. Hourly variation of relative humidity and ambient air temperature for a typical year in Hong Kong.

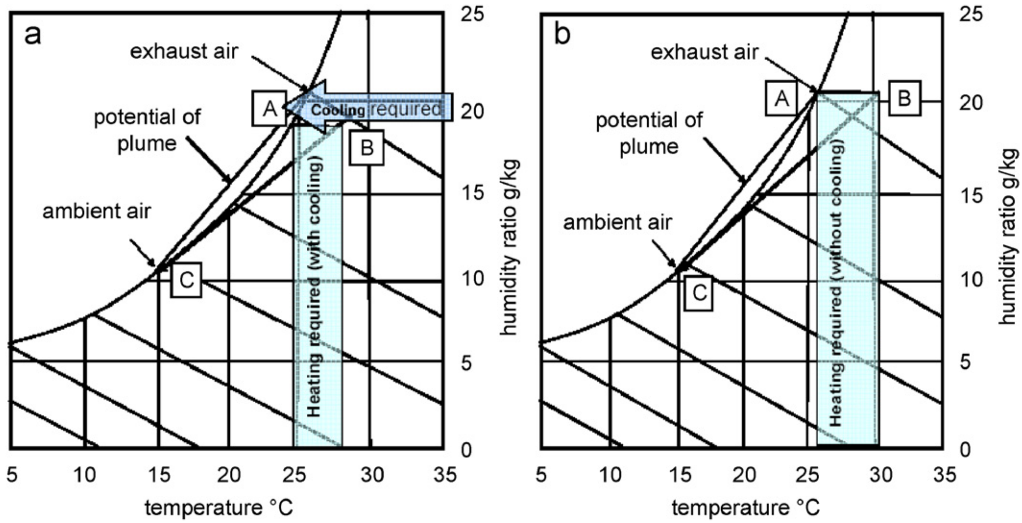

Fig. 3. The heating and cooling requirements for a typical cooling tower (a) with cooling option, (b) without cooling option.

calculated by the models given above. But T_{set} and T_{aset} need to be calculated based on the exhaust and inlet air temperatures of the cooling tower by plotting (on the psychrometric chart) as mentioned in the text and shown in Figs. 3(a) and (b), respectively. Again, based on the total heating requirement, the cooling can be calculated by the ratio of the heating to the cooling (Q_H/Q_L) of the heat pump installed for this purpose. On the other hand, the requirement of the heating can be calculated directly from Fig. 3(b). For further details the readers are encouraged to go through Refs. [27,28].

3. Formation and potential of visible plume

As mentioned above, the air passing through the heat exchanger (fill) comes in the contact of hot water, takes heat and moisture, and normally exits the tower in saturated or superheated state. This plume then gradually remixes with the cooler ambient air and as it cools down the excess moisture condenses in small fog droplets, creating visible plume. This is shown in a simplified psychrometric diagram of Fig. 1(b). During winter season, the cold and humid ambient air at condition 1 is warmed up to condition 2 in the tower (after absorbing the heat released by the hot water) and then remixes with the ambient air along the line 2-1. Most of this mixing occurs in the superheated region, i.e. above the saturation curve, as shown in Fig. 1(b). Where there is larger moisture content in the ambient air, such as in Hong Kong, than that can be held at equilibrium, the condensation occurs and a dense and persistent plume formed. On the other hand, during the hot and dry weather conditions, the ambient air may be heated and remixes along the line 4-3, which is wholly in the sub-saturated region and no visible plume is formed. As mentioned above, the exhaust of the tower is generally saturated and during odd weather conditions cannot be fully absorbed by the surrounding air, as a result, the access appears as visible fog.

As shown in Fig. 1(b), larger the temperature difference between inlet and outlet air of the tower (i.e. the greater the area of the intersection to the left of the saturation curve) more intense will be the plume. Since, the exhaust of the tower is almost saturated, the main parameter causing the visible plume is the temperature difference between the inlet and outlet air of the tower. Also as can be seen from Fig. 1(c), the intensity of plume is more for lower ambient temperature and vice-versa. The area between the straight line and the saturation curve in the superheated region predicts the possibility of the plume, Fig. 1(c). Thus by joining these two points by a straight line one can predict the possibility and the potential of the visible plume, based on the area of intersection. The area of intersection can be evaluated using some standard mathematical formulation with the help of definite integral, as:

$$A = 2\int_{w_2}^{w_1} \sqrt{w}\,\mathrm{d}s, \tag{17}$$

where w_1 and w_2 represent the concentration (presence) of water in the air (g/kg) and $\sqrt{w}$ represents the curve shown in Fig. 1(c). The area of intersection given by Eq. (17) represents the magnitude of the potential of the visible plume based on the inlet and outlet air temperatures of the tower, but has nothing to do to control it. For the critical regions marked on Fig. 2, it is found that the winter conditions have more potential and frequency of plume than those of the spring conditions. Also for these conditions are mentioned in Table 1 and marked on Fig. 1(d).

Table 1
Selected critical weather data for Hong Kong in winter and spring

Winter case		Spring case	
Temperature range (°C)	Relative humidity range (%)	Temperature range (°C)	Relative humidity range (%)
10–16	90–98	20–25	90–98

Using standard simulation technique [26–28] at these particular weather conditions it is found that only three chillers are required to fulfill the cooling requirement of this building. Based on the temperature range of winter condition given in Table 1, we have chosen the ambient temperature as 15 °C and the relative humidity as 95%. The optimum operating parameters such as the building load, cooling load, heating load, dry and wet bulb temperatures, inlet air temperature, relative humidity, water flow rate, air flow rate are similar to those given by earlier authors [26–28]. Again, for different operating conditions the outlet temperatures of the high- and low-speed tower is different and basically this depends on the number and speed of the cooling towers. As mentioned by earlier authors [26–28], the potential of visible plume is found to be higher for low-speed tower which is reverse in the case of high-speed tower. This is because of the fact that the mixing of air (volume flow rate) in the high-speed tower is much higher than that of the low-speed tower and hence, the exhaust temperature of a high-speed tower is much less than that of a low-speed tower and vice-versa. So there is big difference between the outlet temperature of the high-speed and the low-speed cooling towers.

There are two different ways to utilize these heat pumps; one is to use the cooling capacity to cool down the exhaust of the towers and then heat it up, second is just to heat up the exhaust without bothering about the cooling produced by the heat pump. As pointed out by Wang and Tyagi [26] and Tyagi et al. [27] that the former case is more effective provided the optimum distribution of the exhaust for cooling and heating is maintained. As far as heat pumps are concerned, it was found that from energy conservation point of view, the combination of heating and cooling is a better choice. Again, since solar energy is clean, free of cost, and abundantly available on the earth it can be harnessed efficiently, provided some advance techniques are used. Wang et al. [28] applied the heating option using typical flat plate solar collector [29] with two different circulating fluids, one as air and the other as water. Thus the present work is a combined study based on the economic considerations and cost comparisons for the use of different heat pumps and solar collector systems at a typical weather condition.

Based on the literature, the heating and cooling requirements, power consumption, the investment and operational costs, etc. are calculated. The economic considerations and cost comparisons for the use of heat pumps and solar collectors are given in detail at a particular building for a typical weather condition in Hong Kong. The data for different options are shown in Tables 2–5 and the discussion of results is as given below:

4. Discussion of results

As the combined heating and cooling option is possible only in the case of heat pump systems, the solar collector cannot be used for dual application. Also as the heat pump systems and the solar collector systems are two different modes of a real-life application, the details of both the systems are given separately. For various cases the requirements of heating, the consumption of power in different sub-systems, the investment cost, running cost, and other cost are shown in Table 2 for water- and air-cooled heat pump systems, respectively. Similarly, the requirements of heating, power consumption, investment, and operational (running and other) costs are shown in Table 3 for the electric and the geothermal (water- and air-cooled) heat pump systems, respectively. The specification of solar collectors and the investments cost are shown in Table 4, whereas, for the water- and air-cooled solar collectors, the temperature difference, number of solar collectors, mass

Table 2
Comparison between costs and power consumptions for water and air cooled heat pump systems

Heating capacity (kW)	Consumption of power (kW)				Different costs (in US$)					
	Heat pump		Chiller	Fans[a]	Capital		Running		Other	
	Water	Air			Water	Air	Water	Air	Water	Air
1988.8	390.0	641.5	3486.5	270	119,807	215653	520	855	5990	10783
1311.2	257.1	423.0	2651.4	450	78,988	142178	343	564	3949	7109
963.6	188.9	310.8	2834.1	270	58,048	104487	252	414	2902	5224
917.7	179.9	296.0	2633.7	300	55,283	99510	240	395	2764	4975
786.6	154.2	253.7	2651.4	450	47,386	85294	206	338	2369	4265
655.5	128.5	211.5	2462.7	600	39,488	71078	171	282	1974	3554
563.7	110.5	181.8	2404.2	900	33,958	61124	147	242	1698	3056
491.7	96.4	158.6	2462.7	600	29,620	53317	128	211	1481	2666

[a]Fan power consumption depends on the speed and number of cooling towers in use.

Table 3
Comparison among various costs between electric and geothermal heat pump systems

Heating capacity (kW)	Investment cost (US$)				Operational cost (US$)							
	EHP		GHP		Running cost[a]				Other cost[b]			
	Water	Air	Water	Air	EHP		GHP		EHP		GHP	
					Air	Water	Air	Water	Water	Air	Water	Air
1988.8	413,017	442519	531,022	590025	688	597	533	394	12,391	13276	15,931	17701
1311.2	272,299	291749	350,099	388999	453	394	352	260	8169	8752	10,503	11670
963.6	200,112	214406	257,287	285875	333	289	258	191	6003	6432	7719	8576
917.7	190,580	204193	245,032	272258	317	275	246	182	5717	6126	7351	8168
786.6	163,355	175023	210,027	233364	272	236	211	156	4901	5251	6301	7001
655.5	136,129	145852	175,023	194470	226	197	176	130	4084	4376	5251	5834
563.7	117,065	125426	150,512	167235	195	169	151	112	3512	3763	4515	5017
491.7	102,112	109406	131,287	145875	170	148	132	98	3063	3282	3939	4376

[a]Running cost is calculated for per day.
[b]Other cost is considered to be 3–5% (annually) of the capital cost.

flow rates, and the power consumption are shown in Tables 5 and 6, respectively. Also the requirements of heating to control the plume are given in first column of each table [Tables 2 and 3] for heat pump systems while there are only three different cases as given in different rows in Tables 5 and 6 for solar collector systems.

It is also important to note that as the heat pump case is dual, the heating requirement may vary unlike the solar collector case. Also there are both heating and cooling as well as heating alone options for heat pump systems unlike the solar collector systems, that is why Tables 2 and 3 indicate more options than those given in Tables 5 and 6. For solar collector system the fuel (the sun light) is free and the investment cost is very less as compared to those of the heat pump systems and hence, the former is much cheaper than the latter. Again, the running cost for the solar collector systems is nominal as compared to those of

Table 4
Specifications of a typical flat plate solar collector

Dimensions	1860 mm (length) × 1240 mm (width)
Weight	48 kg. (approx)
Collector box	Powder-coated extruded Aluminum channel 1.6 mm thick
Absorber panel	Black continuously plated tig welded solchrome solar selectively coated fins and tubes: 10 nos. Fin size: 122 mm (width) × 1700 mm (length); tube size: 12.7 mm (od) × 0.56 mm (thick) × 1715 mm (length)
Header	Copper tubes 25.4 mm (od) × 0.71 mm (thick) × 1320 mm (length)
Glazing	Toughened/tempered glass
Thermal insulation	Resin bonded rock wool

Number of collectors	Investment and maintenance costs[a]
01	Rs. 12,500/-
05	Rs. 60,000/-
10	Rs. 1,20,000/-
15	Rs. 1,80,000/-
20	Rs. 2,40,000/-

[a]The maintenance will be carried out by the supplier for a period of 5 years. For further details, readers are encouraged to go through Ref. [29].

Table 5
The mass flow rates in solar collector systems

Temperature difference (°C)	Number of collectors	Mass flow rate (kg/s)					
		$Q_H = 1988.8$ kW		$Q_H = 1311$ kW		$Q_H = 655.5$ kW	
		Water	Air	Water	Air	Water	Air
10	20	2.37	9.88	1.56	6.52	0.78	3.26
	15	3.16	13.18	2.08	8.59	1.04	4.34
	10	4.74	19.77	3.12	13.03	1.56	6.52
	5	9.47	39.54	6.24	26.06	3.12	13.03
8	20	2.96	12.36	1.95	8.14	0.98	4.07
	15	3.95	16.47	2.60	10.86	1.30	5.43
	10	5.92	24.71	3.90	16.29	1.95	8.14
	5	11.84	49.42	7.80	32.58	3.90	16.29
6	20	3.59	16.47	2.60	10.86	1.30	5.43
	15	5.26	21.97	3.47	14.48	1.73	7.24
	10	7.89	32.95	5.20	21.72	2.60	10.86
	5	15.89	65.90	10.40	43.44	5.60	21.72
4	20	5.92	24.71	3.90	16.29	1.95	8.14
	15	7.89	32.95	5.20	21.72	2.60	10.86
	10	11.84	49.42	7.80	32.58	3.90	16.29
	5	23.68	98.85	15.61	65.16	7.80	32.58
2	20	11.84	49.42	7.80	32.58	3.90	16.29
	15	15.78	65.90	10.40	43.44	5.20	21.72
	10	23.68	98.85	15.61	65.16	7.80	32.58
	5	47.35	197.69	31.21	130.32	15.61	65.16

Table 6
Detail about the energy consumption, investment cost, operational cost, etc. for water and air cooled solar collector systems

Temperature difference (°C)	Consumption of power[a] (kW h)		Investment cost with the maintenance (US$)	Running cost per day (US$)	
	Water cooled	Air cooled		Water cooled	Air cooled
Heating load = 1988.6 *kW*					
10	8.92	20.69	6000 (20[b])	1.49	3.45
8	11.15	25.86	5425 (15[b])	1.86	4.31
6	14.86	34.84	3625 (10[b])	2.48	5.75
4	22.30	51.72	1500 (05[b])	3.72	8.62
2	44.59	103.43		7.43	17.2
Heating load = 1311 *kW*					
10	5.88	13.64	6000 (20[b])	0.98	2.27
8	7.35	17.05	5425 (15[b])	1.22	2.84
6	9.80	22.73	3625 (10[b])	1.63	3.79
4	14.70	34.09	1500 (05[b])	2.45	5.68
2	29.40	68.18		4.90	11.4
Heating load = 655.5 *kW*					
10	2.94	6.82	6000 (20[b])	0.49	1.14
8	3.57	8.52	5425 (15[b])	0.61	1.42
6	4.90	11.36	3625 (10[b])	0.82	1.89
4	7.35	17.05	1500 (05[b])	1.22	2.84
2	14.70	34.09		2.45	5.68

[a]Power consumption is calculated for 8 h per day.
[b]Number of collectors.

different heat pump systems and hence, the former case can be a better option from the point of view of economics as well as from the point of view of thermodynamics. But as we know the solar energy is intermittent in nature and available only in the day time. At the same time, the building is a commercial and hence, the most working hours will be in the day. So the application of solar collectors can be an alternative but may or may not be a substitute provided weather condition is not a constraint.

When the options and results given in Tables 2 and 3 are compared with those given in Tables 5 and 6, it is found that the overall cost of the solar collector system is almost negligible as compared to those of the heat pump systems for all possible cases. Again, the air-cooled systems are found to be more expensive than those of water-cooled systems. It can be seen from Table 2 that almost every cost (investment, running, maintenance, power input) in the case of air-cooled heat pump systems is found to be much higher than those of the water-cooled heat pump systems. This is because of the properties and constraints attached with these two fluids. In other words, in case of water, a pump is used to circulate water and consumes very less power while in the case of air, a compressor is used and consumes more energy as compared to that of a pump. At the same time, the investment as well as maintenance costs of a compressor are much higher than that of a pump and hence, the running as well as the investment cost of an air-cooled systems is much higher than that of a water-cooled system. Again, a similar explanation can be given between a water-cooled solar collector and an air-cooled solar collector.

Also there is a possibility that the exhaust of air-cooled heat pump systems can be re-mixed with the exhaust of the cooling towers while it is not possible in the case of the water-cooled heat pump system. In that way, the cost of a heat exchanger (to be used for water-cooled system) and the cost of energy loss (in the form of energy efficiency of heat exchanger) for such systems can be saved. Again, as the heat transfer processes depend on the medium, water is a better choice and preferred for heating and cooling applications, especially, where size and space are not the constraints. Also for a water-cooled system, continuous (in a closed loop) circulation of water is needed, so water fouling treatment and cleaning and hence, extra care and maintenance are needed.

The results about the costs and use of the electric and the geothermal heat pump systems are shown in Table 3. This table indicates that the investment cost and other cost of an electric heat pump system is lesser than the geothermal heat pump system while it is reverse in the case of the running cost. The running and other costs are different for different systems and fluids, for example, the investment cost and operational costs for water-cooled systems in both cases is lower than those of the air-cooled systems and can be explained based on the reasons given in the above section.

The specification of a typical flat plate solar collector is given in Table 4 along with the cost for number of collectors with similar specifications. As we can see from Tables 5 and 6, it is clear that for larger temperature difference either the number of collectors and/or the mass flow rate of the fluid is smaller while it is reverse in the case of smaller temperature difference. Also for the same specifications (Table 4), the mass flow rate of water is much less than that of the mass flow rate of the air, which is basically due to the large difference in the specific heat and other physical properties of the two fluids. Again for the water-cooled collector there is a need of pump, while it is an air compressor in the case of air-cooled collector which in turn costs more in the form of investment and operational costs. Thus the water-cooled collectors can give a better output provided optimum configuration is used. Again, for water-cooled collectors the heat transfer process needs a heat exchanger and leads to the loss of energy along with the extra cost while there is no need to maintain the quality of air for such applications and hence, the hot air from the collector can be mixed directly with the exhaust of the cooling tower. Thus in the air-cooled collector systems, the direct mixing of the exhaust of the cooling tower at state point 7 and hence, extra cost in the form of heat exchanger as well as the energy loss can be reduced.

As mentioned by earlier workers [26–28] the total power consumption first decreases and then increases while the power consumption in the chillers decreases, whereas, the COP of the chillers increases, as the number of cooling towers increases. On the other hand, the power consumption in the fan which solely depends on the mode of operation of cooling towers increases continuously, except for one combination. Thus the proper arrangement and the optimized operation (number and speed) of cooling towers may be an effective means to control and/or at least to reduce the potential of visible plume generated by wet cooling towers in commercial buildings at Hong Kong and other similar climates around the world. Finally, two fluids and the different options mentioned in this study have their own advantage and disadvantage, and hence, the optimum configuration is a better choice for such application and can be used for the present case.

5. Conclusions

The present work is a detail comparative economic study for the heat pump as well as for solar collector systems to control the visible plume from wet cooling towers of a commercial building in Hong Kong. The study is done and compared using the total cost which consists of investment/capital and operational (running and other) costs, while taking into account some of the constraints. The capital cost is the actual cost of the device while the operational cost is the cost that keeps the system working in good condition. In other words, running cost is the cost that keeps the system working while other cost is the cost that keeps the system healthy and efficient. In layman's language, running cost is basically the cost of the energy and man power that keeps the system working while other cost is basically the maintenance cost. For heat pump, the cost of the input power to the compressor is the running cost and for the solar collector, the running cost is the cost of the power consumed by the pump/blower to circulate the fluid which is very less compared to the former. In this case study, calculations are done, for the water-cooled solar collector as well as for the air-cooled solar collectors. The comparison is also given among the different possible options using the previous work as a base and some of the comparative conclusions are given as below:

1. For all the cases explained in this study, the air-cooled systems are found to be more expensive than those of water-cooled systems. It can be seen from different tables that almost every cost (investment, running, maintenance, power input) in the case of air-cooled heat pump system is found to be much higher than those of the water-cooled heat pump system. This is because of the properties and constraints attached with these two fluids. In other words, in the case of water, a pump is used to circulate water and consumes very less power while in the case of air, a compressor is used. At the same time the investment cost as well as consumption of power in the case of a compressor is much higher than that of a pump and hence, the running cost as well as the investment cost of an air-cooled system is much higher than that of a water-cooled system.
2. Also there is a possibility that the exhaust of air-cooled system can be remixed with the exhaust of the cooling towers while it is not possible in the case of the water-cooled system. In that way, we can also save the investment cost of the heat exchangers as well as energy requirement due to heat loss. Again, as the heat transfer process depends on the medium, water is a better choice and preferred for heating and cooling applications, especially where size and space are not the constraints. For water-cooled system, a continuous (in a closed loop) circulation of water is needed, and water fouling treatment and hence, extra care and maintenance of this system are needed.
3. The results about the costs and use of the electric and the geothermal heat pump systems indicate that the investment cost and other cost of an electric heat pump system is lesser than the geothermal heat pump system while it is reverse in the case of the running cost. The running and other costs are different for different systems and fluids, for example, the investment cost and operational costs for water-cooled systems in both cases is lower than those of the air-cooled systems.
4. For solar collector system, the fuel (the sun light)is free and the investment cost is very less as compared to those of the heat pump systems and hence, the former is much cheaper than the latter. Again, the running cost for the solar collector systems is nominal as compared to those of different heat pump systems and can be a better option

from the point of view of economics as well as from the point of view of thermodynamics. So the application of solar collectors can be an alternative but may not be a substitute provided weather condition is not a big constraint.

5. Finally, the combined heating and cooling option is feasible only in the case of heat pump systems and hence, the solar collector cannot be used for dual application. The combined option not only consumes less energy but also utilizes the cooling produced by the heat pump system and hence, the combined option is more economical than the heating alone option especially in the case of heat pump systems. Also the temperature of the refrigerant is much higher at the exit of the compressor, i.e. at the entering of the condenser. So a desuperheater can be used to utilize the heat produced by the condenser, especially, where there is no need of heating the building such as Hong Kong.

Finally, proper combination and optimum operation of cooling towers can be an effective means to control and/or at least to reduce the potential and frequency of plume generated by cooling towers at Hong Kong and similar climates. Finally, the two fluids and the different options mentioned in this study have their own advantage and disadvantage, and hence, the optimum configuration is a better choice for any application and can be used for the present case.

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